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DESCRIPTION

INTERNAL GEAR TYPE OIL PUMP ROTOR

TECHNICAL FIELD

The present invention relates to an oil pump rotor assembly used in an internal gear type oil pump which draws and discharges fluid by volume change of cells formed between an inner rotor and an outer rotor.

BACKGROUND ART

Conventionally, an internal gear type oil pump includes an outer rotor having internal teeth, an inner rotor having external teeth which are engageable with the internal teeth, and a casing in which a suction port for drawing fluid and a discharge port for discharging fluid are formed. The inner rotor is rotated so that the outer rotor is rotated while the external teeth engage the internal teeth, which produces changes in the volumes of cells formed between the inner rotor and the outer rotor, and thereby fluid is drawn and is discharged.

Each of the cells is independently delimited at a front portion and at a rear portion as viewed in the direction of rotation by the external teeth of the inner rotor and the internal teeth of the outer rotor. The volume of each of the cells is minimized at a rotational position in which one of the tooth tips of the external teeth of the inner rotor positionally coincides with one of the tooth spaces of the internal teeth of the outer rotor, and, from this rotational position, the cell draws fluid as the volume thereof increases while moving over the suction port. The volume of each of the cells is maximized at a rotational position in which one of the tooth spaces of the external teeth of the inner rotor

positionally coincides with one of the tooth spaces of the internal teeth of the outer rotor, and, from this rotational position, the cell discharges fluid as the volume thereof decreases while moving over the discharge port.

In the internal gear type oil pump, the inner rotor is driven so as to rotate, and the outer rotor is rotated because tooth surfaces of the external teeth push tooth surfaces of the internal teeth. Here, the engagement between the rotors, by which rotational force is transmitted, is reviewed. The rotational force is transmitted in the direction substantially perpendicular to the tooth surfaces when the teeth are placed near a position at which the volume of the cell is minimized. On the other hand, when the teeth are placed near a position at which the volume of the cell is maximized, because the tooth tips of the rotors contact each other, the rotational force is not transmitted in the direction substantially perpendicular to the tooth surfaces, and components of slip and friction are dominant.

When the tooth surfaces of the rotors contact each other where slip is dominant, the teeth do not contribute to transmission of the rotational force, and sliding friction is increased due to contact between the teeth, which may lead to operation noise, and decrease in mechanical efficiency.

In order to solve this problem, rotors have been proposed, in each of which a recess is formed in the tooth surface to eliminate contact which does not contribute to transmission of the rotational force (see, for example, Japanese Unexamined Patent Application, First Publication No. Hei 09-166091).

In general, in an internal gear type oil pump rotor assembly as mentioned above, clearances are formed between the tooth surfaces of the rotors, which define a cell. The main reason for providing such clearances is to prevent problems in which rotation of the rotors becomes impossible or noise is emitted because the tooth tips of the rotors

interfere with each other due to undesirable shapes and accuracy of assembly of the rotors, and practical countermeasures have been proposed such that the profiles of the teeth of the outer rotor are uniformly cut, the curve defining the shape of the teeth is partially flattened, or the like.

However, when such clearances are merely provided by taking conventional measures such as uniform cut of the tooth profiles, partial flattening of the tooth surface, or providing the recess, backlash between the teeth is unnecessarily increased; therefore, another problem is encountered in that it is difficult to prevent noise due to irregular oscillation of the rotors during rotation.

DISCLOSURE OF THE INVENTION

The present invention was conceived in view of the above circumstances, and an object of the present invention is to provide an internal gear type oil pump rotor assembly which stably rotates without emitting excessive noise.

In order to achieve the above object, the present invention provides an oil pump rotor assembly including: an inner rotor having "n" external teeth ("n" is a natural number); and an outer rotor having (n+1) internal teeth which are engageable with the external teeth, wherein the oil pump rotor assembly is used in an oil pump which, during rotation of the inner and outer rotors, draws and discharges fluid by volume change of cells formed between the inner rotor and the outer rotor, wherein when a clearance, which is defined between the teeth of the inner and outer rotors that together form one of the cells which has the minimum volume among the cells, is designated as "a", a clearance, which is defined between the teeth of the inner and outer rotors that together form one of the cells whose volume is increasing during rotation of the inner and outer rotors, is designated as "b", and a clearance, which is defined between the teeth of the

inner and outer rotors that together form one of the cells which has the maximum volume among the cells, is designated as "c", the following inequalities are satisfied:

$a \leq b \leq c$, and $a < c$,

and wherein when the clearance "b" in the cell positioned forward as viewed in the direction of rotation is further designated as "b1", and the clearance "b" in the cell positioned backward as viewed in the direction of rotation is further designated as "b2", the following inequality is satisfied:

$b1 \leq b2$.

In the above oil pump rotor assembly, when a clearance, which is defined between the teeth of the inner and outer rotors that together form one of the cells whose volume is decreasing during rotation of the inner and outer rotors, is designated as "d", the following inequalities are satisfied:

$a \leq b \leq c$, $a < c$, and $a \leq d \leq c$,

and when the clearance "d" in the cell positioned backward as viewed in the direction of rotation is further designated as "d1", and the clearance "d" in the cell positioned forward as viewed in the direction of rotation is further designated as "d2", the following inequality is satisfied:

$d1 \geq d2$.

The present invention further provides an oil pump rotor assembly including: an inner rotor having "n" external teeth ("n" is a natural number); and an outer rotor having (n+1) internal teeth which are engageable with the external teeth, wherein the oil pump rotor assembly is used in an oil pump which, during rotation of the inner and outer rotors, draws and discharges fluid by volume change of cells formed between the inner rotor and the outer rotor, and wherein a clearance, which is defined between the teeth of the inner and outer rotors that together form one of the cells, gradually increases as the cell

rotationally moves from a position at which the volume of the cell is minimized to a position at which the volume of the cell is maximized.

In the above oil pump rotor assembly, the clearance, which is defined between the teeth of the inner and outer rotors that together form one of the cells, may gradually decrease as the cell rotationally moves from a position at which the volume of the cell is maximized to a position at which the volume of the cell is minimized.

According to these inventions, because the clearance between the rotors that together form the cell is minimized at an engagement region, and then the clearance is continuously increased, without decreasing, to a maximum size, backlash at a position at which the teeth engage each other is minimized, and a sufficient clearance is ensured at a rotational position at which the teeth do not contribute to engagement. The external teeth engage the internal teeth at a position at which a slip component is minimized so as to transmit rotational force, and the external teeth and the internal teeth do not contribute to transmitting rotational force at a position at which a slip component is increased. Therefore, an internal gear type oil pump rotor assembly can be obtained which does not emit excessive noise while having low levels of friction and high mechanical efficiency.

Moreover, because in the process in which the volume of the cell is decreasing, the clearance between the rotors gradually decreases, without increasing, to a minimum size, a sufficient clearance is ensured where the teeth do not contribute to engagement while minimizing backlash where the teeth engage each other, and thus an internal gear type oil pump rotor assembly can be obtained which does not emit excessive noise while having low levels of friction.

In the above oil pump rotor assembly, the tooth surfaces of the inner and outer rotors may be respectively formed using cycloid curves which are formed by rolling respective rolling circles along respective base circles without slip.

In the above oil pump rotor assembly, the tooth surfaces of the inner rotor may be formed using a trochoid envelope curve which is formed by moving a trajectory circle, whose center is positioned on a trochoid curve, along the trochoid curve, and the tooth tips of the outer rotor may be formed using an arc having the same radius as that of the trajectory circle.

According to these inventions, a cycloid type rotor assembly which is formed using cycloid curves and a trochoid type rotor assembly which is formed using trochoid curves, both of which have been conventionally used, can be made so as to emit less noise and to have lower levels of friction.

In the above oil pump rotor assembly, each of the tooth profiles of the inner rotor may be formed such that the tip profile thereof is formed using an epicycloid curve which is formed by rolling a first circumscribed-rolling circle Ai along a base circle Di without slip, and the tooth space profile thereof is formed using a hypocycloid curve which is formed by rolling a first inscribed-rolling circle Bi along the base circle Di without slip, and each of the tooth profiles of the outer rotor is formed such that the tip profile thereof is formed using an epicycloid curve which is formed by rolling a second circumscribed-rolling circle Ao along a base circle Do without slip, and the tip profile thereof is formed using a hypocycloid curve which is formed by rolling a second inscribed-rolling circle Bo along the base circle Do without slip, and the inner rotor and the outer rotor may be formed such that the following equations are satisfied:

$øBo=øBi$;

$øDo=øDi\cdot(n+1)/n+t\cdot(n+1)/(n+2)$; and

$øAo=øAi+t/(n+2)$,

where øDi is the diameter of the base circle Di of the inner rotor, øAi is the diameter of the first circumscribed-rolling circle Ao, øBi is the diameter of the first inscribed-rolling

circle Bi, $øDo$ is the diameter of the base circle Do of the outer rotor, $øAo$ is the diameter of the second circumscribed-rolling circle Ao, $øBo$ is the diameter of the second inscribed-rolling circle Bo, and t ($\neq 0$) is a clearance between the tooth tip of the inner rotor and the tooth tip of the outer rotor.

In this case, when tooth profiles of the inner and outer rotors are determined, because the sum of the rolling distances of the circumscribed-rolling circle and the inscribed-rolling circle of the inner rotor must be equal to the circumferential length of the base circle thereof, and the sum of the rolling distances of the circumscribed-rolling circle and the inscribed-rolling circle of the outer rotor must be equal to the circumferential length of the base circle thereof, the following equations must be satisfied:

$øDi=n\cdot(øAo+øBo)$; and

$øDo=(n+1)\cdot(øAo+øBo)$.

In addition, in this configuration, the diameters of the inscribed-rolling circles of the inner and outer rotors are set to be the same with respect to each other, i.e.,

$øBo=øBi$

in order to reduce the circumferential clearance between the tooth space of the inner rotor and the tooth tip of the outer rotor.

The diameter of the base circle of the outer rotor is greater than in the case of a conventional oil pump rotor assembly, i.e.,

$øDo=øDi\cdot(n+1)/n+(n+1)\cdot t/(n+2)$.

Because the total of a multiple of the rolling distance of the circumscribed-rolling circle and a multiple of the rolling distance of the inscribed-rolling circle must agree with the length of circumference of a base circle, the diameter of the circumscribed-rolling circle of the outer rotor must be adjusted as follows:

$øAo=øAi+t/(n+2)$.

According to this oil pump rotor assembly, because an appropriate radial clearance is ensured between the external teeth of the inner rotor and the internal teeth of the outer rotor, and the circumferential clearances between the teeth of the rotors are reduced from that in the conventional case, rattling generated between the rotors is reduced, and quietness of the oil pump can be improved.

As another configuration of an oil pump rotor assembly, each of the tooth profiles of the inner rotor may be formed such that the tip profile thereof is formed using an epicycloid curve which is formed by rolling a first circumscribed-rolling circle Di along a base circle "bi" without slip, and the tooth space profile thereof is formed using a hypocycloid curve which is formed by rolling a first inscribed-rolling circle "di" along the base circle "bi" without slip, and each of the tooth profiles of the outer rotor is formed such that the tip profile thereof is formed using an epicycloid curve which is formed by rolling a second circumscribed-rolling circle Do along a base circle "bo" without slip, and the tip profile thereof is formed using a hypocycloid curve which is formed by rolling a second inscribed-rolling circle "do" along the base circle "bo" without slip, and the inner rotor and the outer rotor may be formed such that the following equations and inequalities are satisfied:

$øbi=n \cdot (øDi+ødi)$;

$øbo=(n+1) \cdot (øDo+ødo)$;

one of $øDi+ødi=2e$ and $øDo+ødo=2e$;

$øDo > øDi$;

$ødi > ødo$; and

$(øDi+ødi) < (øDo+ødo)$,

where øbi is the diameter of the base circle "bi" of the inner rotor, øDi is the diameter of

the first circumscribed-rolling circle Di, ødi is the diameter of the first inscribed-rolling circle "di", øbo is the diameter of the base circle "bo" of the outer rotor, øDo is the diameter of the second circumscribed-rolling circle Do, ødo is the diameter of the second inscribed-rolling circle "do", and "e" is an eccentricity distance between the inner and outer rotors.

In this case, when tooth profiles of the inner and outer rotors are determined, because the sum of the rolling distances of the circumscribed-rolling circle and the inscribed-rolling circle of the inner rotor must be equal to the circumferential length of the base circle thereof, and the sum of the rolling distances of the circumscribed-rolling circle and the inscribed-rolling circle of the outer rotor must be equal to the circumferential length of the base circle thereof, the following equations must be satisfied:

$øbi=n\cdot(øDi+ødi)$; and

$øbo=(n+1)\cdot(øDo+ødo)$.

The tooth tip profile of the inner rotor which is formed by the first circumscribed-rolling circle Di with respect the tooth space profile of the outer rotor which is formed by the second circumscribed-rolling circle Do, and the tooth tip profile of the outer rotor which is formed by the second inscribed-rolling circle "do" with respect to the tooth space profile of the inner rotor which is formed by the first inscribed-rolling circle "di" are determined such that the following inequalities are satisfied:

$øDo > øDi$; and

$ødi > ødo$,

so that a large backlash, which is defined between the tooth surfaces of the rotors during engagement, is ensured. Here, the backlash is a gap formed between the tooth surface

of the inner rotor, which is opposite to the tooth surface to which force is applied during engagement, and the tooth surface of the outer rotor.

Moreover, because the inner rotor and the outer rotor engage each other, one of the following equations must be satisfied:

$øDi+ødi=2e$; and

$øDo+ødo=2e$.

Furthermore, in this invention, in order to make the inner rotor smoothly rotate in the outer rotor while ensuring tip clearance and an appropriate size of backlash, and reducing an engagement resistance, the diameter of the base circle of the outer rotor is made greater than that in a conventional case so that the base circle of the inner rotor does not contact the base circle of the outer rotor at the engagement region at which the inner rotor engages the outer rotor, i.e., the following inequality is satisfied:

$(n+1)\cdot øbi < n\cdot øbo$.

Accordingly, the following inequality is derived:

$(øDi+ødi) < (øDo+ødo)$.

According to the above configuration, because circumferential clearances (along the circumference of the base circle) between the tooth surfaces of the rotors are made smaller than in conventional cases while ensuring tip clearances between the external teeth of the inner rotor and the internal teeth of the outer rotor, play between the rotors can be reduced, and a quiet oil pump can be made. Specifically, impacts between the internal teeth of the outer rotor and external teeth of the inner rotor can be prevented even when driving torque for the oil pump rotor assembly changes while oil pressure in the oil pump rotor assembly is low; therefore, quietness of the oil pump rotor assembly can be ensured.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a plan view of an internal gear type oil pump rotor assembly according to a first embodiment of the present invention, in which inter-tooth clearances “a”, “b”, and “d” are shown.

FIG. 2 is a plan view of the internal gear type oil pump rotor assembly according to the first embodiment of the present invention, in which an inter-tooth clearance “c” is shown.

FIG. 3 is a graph in which the inter-tooth clearance of the internal gear type oil pump rotor assembly of the present invention shown in FIG. 1 and that of a conventional rotor assembly are compared, with respect to the rotational angle of the inner rotor.

FIG. 4 is a plan view showing an oil pump rotor assembly according to a first embodiment of the present invention in which the inner and outer rotors thereof satisfy the following equations:

$øBo=øBi$;

$øDo=øDi\cdot(n+1)/n+t\cdot(n+1)/(n+2)$; and

$øAo=øAi+t/(n+2)$,

and t is set to be 0.12 mm.

FIG. 5 is an enlarged view showing the engagement region, indicated by V, of the oil pump shown in FIG. 4.

FIG. 6 is a graph showing comparison between noise from the oil pump incorporating the oil pump rotor assembly shown in FIG. 4 and noise from a conventional oil pump.

FIG. 7 is a plan view showing a third embodiment of the oil pump rotor assembly according to the present invention.

FIG. 8 is an enlarged view showing the engagement region, indicated by VIII, of

the oil pump shown in FIG. 7.

FIG. 9 is a graph showing comparison between a backlash of an oil pump incorporating the oil pump rotor assembly shown in FIG. 7 and a backlash of a conventional oil pump.

FIG. 10 is a graph showing comparison between noise from an oil pump incorporating the oil pump rotor assembly shown in FIG. 7 and noise from a conventional oil pump.

BEST MODE FOR CARRYING OUT THE INVENTION

A first embodiment of the present invention will be explained below with reference to FIGS. 1 to 3.

The internal gear type oil pump rotor assembly shown in FIGS. 1 and 2 is a cycloid type rotor assembly in which teeth of an outer rotor 10 and teeth of an inner rotor 20 are formed using respective cycloid curves, each of which is formed by rolling a rolling circle along a base circle. The parameters of the rotors 10 and 20 are set as follows:

the diameter of the base circle Do of the outer rotor 10 is 57.31 mm;

the diameter of the circumscribed-rolling circle Ao of the outer rotor 10 is 2.51 mm;

the diameter of the inscribed-rolling circle Bo of the outer rotor 10 is 2.70 mm;

the number of teeth Zo of the outer rotor 10 is 11 (teeth);

the diameter of the base circle Di of the inner rotor 20 is 52.00 mm;

the diameter of the circumscribed-rolling circle Ai of the inner rotor 20 is 2.50 mm;

the diameter of the inscribed-rolling circle Bi of the inner rotor 20 is 2.76 mm;

the number of teeth Zi of the inner rotor 20 is 10 (teeth); and

an eccentricity distance "e" is 2.60 mm.

The inner rotor 20 is inscribed in the outer rotor 10 while the external teeth of the inner rotor 20 engage the internal teeth of the outer rotor 10 so as to form cells R between the teeth. Each of the cells R rotationally moves while the volume thereof changes when the inner rotor 20 along with the outer rotor rotate in the direction indicated by the arrows in FIGS. 1 and 2 (in the counterclockwise direction).

When the rotational position θ of the inner rotor 20 is designated as 0° at the bottom of the drawing, and is designated as 180° at the top of the drawing, the volume of each of the cells R gradually increases, as the inner rotor 20 rotates, from a position at which θ=0° (FIG. 1) and the volume thereof is minimized (Vmin), to a position at which θ=198° (FIG. 2) and the volume thereof is maximized (Vmax). Each of the cells R draws fluid through a suction port formed in a casing (not shown) during the process in which the volume of the cell R increases.

Here, an inter-tooth clearance is defined as the region which closes one of the cells R in the circumferential direction, i.e., the region at which the gap between the teeth of the rotors 10 and 20 that together form the cell R is minimized.

When an inter-tooth clearance, which is defined between the teeth of the rotors 10 and 20 that together form one of the cells R which has the minimum volume (Vmin) among the cells, is designated as "a", an inter-tooth clearance, which is defined between the teeth of the rotors 10 and 20 that together form one of the cells R whose volume is increasing during rotation of the rotors 10 and 20, is designated as "b" (FIG. 1), and an inter-tooth clearance, which is defined between the teeth of the rotors 10 and 20 that together form one of the cells R which has the maximum volume (Vmax) among the cells, is designated as "c" (FIG. 2), the following inequalities are satisfied:

$a \leq b \leq c$, and $a < c$.

Moreover, when an inter-tooth clearance, which is defined between the teeth of

the rotors 10 and 20 that together form one of the cells R whose volume is decreasing during rotation of the rotors 10 and 20, is designated as "d", the following inequalities are satisfied:

$a \leq d \leq c$.

The comparison between the clearance between the outer rotor 10 and the inner rotor 20 in the internal gear type oil pump rotor assembly of the present embodiment and that between the rotors in a conventional rotor assembly is shown in FIG. 3.

The clearance in the conventional rotor assembly is maximized where the volume of the cell is minimized, gradually decreases as the cell rotates, and is minimized where the volume of the cell is maximized. Accordingly, in the conventional rotor assembly, the teeth of the rotors tend to contact each other even in zones β and γ in which the clearance is smaller than that in an engagement effect zone α; therefore, due to friction, mechanical efficiency may be decreased, and excessive noise may be emitted.

On the other hand, in the case of the present embodiment, the inter-tooth clearance between the rotors that together form the cell R gradually and continuously increases during the process in which the volume of the cell R increases from the minimum volume (Vmin) to the maximum volume (Vmax), as shown in FIG. 3. More specifically, with regard to the clearance "b" in a range $0° < \theta < 198°$, when the clearance "b" in the cell R positioned forward as viewed in the direction of rotation is further designated as "b1", and the clearance "b" in the cell R positioned backward as viewed in the direction of rotation is further designated as "b2", the following inequality is satisfied over the entire range of the rotational position θ:

$b1 \leq b2$.

When the inner rotor 20 rotates from the rotational position $\theta=0°$, the teeth of the outer rotor 10 and the teeth of the inner rotor 20 engage each other so as to transmit a

rotational force in the zone α shown in FIG. 1. In the zone α (i.e., the engagement effect zone), the clearance continuously increases as shown in FIG. 3, i.e., the clearance in the cell R positioned forward as viewed in the direction of rotation is always greater than that in the cell R positioned backward.

The clearance in the zone β in which the inner rotor 20 has further rotated is greater than that in the zone α, and the clearance increases further. Accordingly, the teeth of the rotors 10 and 20 tend not to contact each other in the zone β when compared with the engagement effect zone α.

The clearance in the zone γ (i.e., a performance effect zone) in which the inner rotor 20 has further rotated is greater than that in the zone β, and the clearance further increases, in accordance with rotation, to a maximum value at the rotational position of the inner rotor θ=198°. Accordingly, the teeth of the rotors 10 and 20 tend not to contact each other in the zone γ when compared with the zone β.

The clearance "c" (FIG. 2), which is the clearance when the volume of the cell R is maximized (Vmax), may affect the performance of the pump because the cell R is at a transition point from drawing to discharging, and the clearance "c" is substantially the same as that in the conventional rotor assembly; therefore, performance of the pump is not degraded.

With regard to the clearance "d" (FIG. 1) in the cell R which is forwarded from the cell R having the maximum volume (Vmax), the clearance "d" gradually decreases, in accordance with the rotation of the inner rotor 20, to a minimum value at the rotational position of the inner rotor θ=396°. In other words, with regard to the clearance "d" in the range of $198° < \theta < 396°$, when the clearance "d" in the cell positioned backward as viewed in the direction of rotation is further designated as "d1", and the clearance "d" in the cell positioned forward as viewed in the direction of rotation is further designated as

"d2", the following inequality is satisfied over the entire range of the rotational position θ:

$d1 \geq d2$.

Accordingly, in the process in which the volume of the cell R decreases, as in the process in which the volume of the cell R increases, the teeth tend not to contact each other in the performance effect zone γ when compared with the engagement effect zone α.

As explained above, in the internal gear type oil pump rotor assembly of the present embodiment, the clearance is made small in the engagement effect zone α in which the rotational force is efficiently transmitted, the clearance is made large in the performance effect zone γ in which the rotational force cannot be efficiently transmitted, and the clearance is made to gradually increase between the zones α and γ; therefore, the rotational force is transmitted by the contact between the teeth mainly in the engagement effect zone α, and the teeth tend not to contact each other in other zones. As a result, excessive noise and degradation of mechanical efficiency can be prevented.

When the clearance is increased from "a" to "c", it is more preferable that inequalities $a<b$, $b1<b2$, and $b<c$ be satisfied; however, conditions in which equations $a=b$, $b1=b2$, or $b=c$ are partially satisfied may be acceptable as long as an inequality $a<c$ is satisfied, i.e., the clearance does not decrease.

Similarly, when the clearance is decreased from "c" to "a", it is more preferable that inequalities $c>d$, $d1>d2$, and $d>a$ be satisfied; however, conditions in which equations $a=b$, $b1=b2$, or $b=c$ are partially satisfied may be acceptable as long as an inequality $c>a$ is satisfied, i.e., the clearance does not increase.

In the oil pump rotor assembly of the present embodiment having the aforementioned dimensions, or in the oil pump rotor assembly having dimensions similar

to these, it is preferable that the value “a” be in the following range:

0.010 mm $\leq$ a $\leq$ 0.040 mm.

When the value “a” is set to be smaller than 0.010 mm, the oil pump rotor assembly may not rotate smoothly, and the function as a pump may be lost. In contrast, when the value “a” is set to be greater than 0.040 mm, backlash may become large, and operation noise may not be reduced.

Moreover, it is preferable that the value “c” be in the following range:

0.040 mm $\leq$ a $\leq$ 0.150 mm.

When the value “c” is set to be smaller than 0.040 mm, engagement in the engagement region (at 0° in FIG. 1) may become impossible. In contrast, when the value “c is set to be greater than 0.150 mm, oil excessively leaks through the gap between the teeth, and discharge performance of the pump will be extremely degraded.

Next, a second embodiment of the present invention will be explained below with reference to FIGS. 4 to 6.

The oil pump rotor assembly shown in FIG. 4 includes an inner rotor 110 provided with “n” external teeth (“n” indicates a natural number, and n=10 in this embodiment), and an outer rotor 120 provided with “n+1” internal teeth (n+1=11 in this embodiment) which are engageable with the external teeth. The inner rotor 110 and the outer rotor 120 are accommodated in a casing 150.

Between the tooth surfaces of the inner rotor 110 and outer rotor 120, there are formed a plurality of cells C in the direction of rotation of the inner rotor 110 and outer rotor 120. Each of the cells C is delimited at a front portion and at a rear portion as viewed in the direction of rotation of the inner rotor 110 and outer rotor 120 by contact regions between the external teeth 111 of the inner rotor 110 and the internal teeth 121 of

the outer rotor 120, and is also delimited at either side portions by the casing 150, so that an independent fluid conveying chamber is formed. Each of the cells C moves while the inner rotor 110 and outer rotor 120 rotate, and the volume of each of the cells C cyclically increases and decreases so as to complete one cycle in a rotation.

The inner rotor 110 is mounted on a rotational axis so as to be rotatable about an axis Oi. Each of the tooth profiles of the inner rotor 110 is formed such that the tooth tip profile thereof is formed using an epicycloid curve which is formed by rolling a first circumscribed-rolling circle Ai along a base circle Di of the inner rotor 110 without slip, and the tooth space profile thereof is formed using a hypocycloid curve which is formed by rolling a first inscribed-rolling circle Bi along the base circle Di without slip.

The outer rotor 120 is mounted so as to be rotatable, in the casing 150, about an axis Oo which is disposed so as to have an offset (the eccentricity distance is "e") from the axis Oi. Each of the tooth profiles of the outer rotor 120 is formed such that the tooth space profile thereof is formed using an epicycloid curve which is formed by rolling a second circumscribed-rolling circle Ao along a base circle Do of the outer rotor 120 without slip, and the tooth tip profile thereof is formed using a hypocycloid curve which is formed by rolling a second inscribed-rolling circle Bo along the base circle Do without slip.

When the diameter of the base circle Di of the inner rotor 110, the diameter of the first circumscribed-rolling circle Ai, the diameter of the first inscribed-rolling circle Bi, the diameter of the base circle Do of the outer rotor 120, the diameter of the second circumscribed-rolling circle Ao, and the diameter of the second inscribed-rolling circle Bo are assumed to be øDi, øAi, øBi, øDo, øAo, and øBo, respectively, the equations which will be discussed below must be satisfied between the inner rotor 110 and the outer rotor 120. Note that dimensions will be expressed in millimeters.

First, with regard to the inner rotor 110, because the total of a multiple of the rolling distance of the first circumscribed-rolling circle Ai and a multiple of the rolling distance of the first inscribed-rolling circle Bi must agree with the length of circumference of a base circle, i.e., the length of circumference of the base circle Di of the inner rotor 110 must be equal to the length obtained by multiplying the sum of the rolling distance per revolution of the first circumscribed-rolling circle Ai and the rolling distance of the first inscribed-rolling circle Bi by an integer (i.e., by the number of teeth of the inner rotor 110),

$\pi \cdot øDi = n \cdot \pi \cdot (øAi + øBi)$, i.e.,

$øDi = n \cdot (øAi + øBi)$ ··· (Ia).

Similarly, with regard to outer rotor 120, the length of circumference of the base circle Do of the outer rotor 120 must be equal to the length obtained by multiplying the sum of the rolling distance per revolution of the second circumscribed-rolling circle Ao and the rolling distance of the second inscribed-rolling circle Bo by an integer (i.e., by the number of teeth of the outer rotor 120),

$\pi \cdot øDo = (n+1) \cdot \pi \cdot (øAo + øBo)$, i.e.,

$øDo = (n+1) \cdot (øAo + øBo)$ ··· (Ib).

Next, the conditions required for determining tooth profiles of the outer rotor 120 according to this embodiment will be explained below based on a conventional outer rotor "ro" (specifically, the second circumscribed-rolling circle "ao" (whose diameter is øao), the second inscribed-rolling circle "bo" (whose diameter is øbo), and the base circle "do"(whose diameter is ødo)).

The outer rotor "ro" engages the inner rotor 110 according to the present

embodiment with a clearance of "t" while being disposed with respect to the inner rotor 110 so as to have an offset (the eccentricity distance is "e"). The clearance "t" is a gap formed between one of the tooth tips of the inner rotor 110 and one of the tooth tips of the outer rotor 120 at a position which is away from an engagement region by 180° along the direction of rotation when the inner rotor 110 and the outer rotor 120 are disposed such that one of the tooth tips of the inner rotor 110 directly contacts one of the tooth spaces of the outer rotor 120 in the engagement region.

Here, the following equations are satisfied:

$\varnothing do=\varnothing Di\cdot(n+1)/n \cdots$ (II);

$\varnothing do=(n+1)\cdot(\varnothing ao+\varnothing bo) \cdots$ (III);

$\varnothing ao=\varnothing Ai+t/2 \cdots$ (IIIa); and

$\varnothing bo=\varnothing Bi-t/2 \cdots$ (IIIb).

The inner rotor 110 engaging the outer rotor "ro" satisfies the following generic equations:

$\varnothing ai+\varnothing bi=\varnothing Ai+\varnothing Bi=2e \cdots$ (1); and

$\varnothing Di=\varnothing do-2e \cdots$ (2).

In this embodiment, in order to decrease the circumferential clearances t2 while ensuring the radial clearance t1 between the tooth tip of the outer rotor 120 and the tooth space of the inner rotor 110 in the engagement region, the diameters are set as follows:

$\varnothing Bo=\varnothing bi=\varnothing Bi \cdots$ (IV).

Based on the above equations (IV) and (1),

$\varnothing ai=\varnothing Ai \cdots$ (3).

When the inscribed-rolling circle of the outer rotor 120 is set as described above, the clearance "t" which is expressed as

$t=(\varnothing Do-\varnothing Bo+\varnothing Ao)-(\varnothing Di+\varnothing Ai+\varnothing Ai)$ can be expressed, using the above equations (1) to (3)

and (IV), as follows:

$t=(øDo-ødo)+(øAo-øai) \cdots (V)$.

Based on the above equations (Ib), (III), (IV), and (V),

$t=(øAo-øai)\cdot(n+2) \cdots (VI)$; therefore,

$øAo=øai+t/(n+2)$.

Next, the diameter øDo of the base circle Do is to be found. Based on the above equations (Ib) and (III),

$øDo-ødo=(n+1)\cdot(øAo+øBo)-(n+1)\cdot(øao+øbo)$.

Furthermore, based on the above equations (IIIa), (IIIb), and (IV),

$øDo-ødo=(n+1)\cdot(øAo-øai) \cdots (VII)$.

By using the equation (VI), the equation (VII) can be expressed as follows:

$øDo-ødo=(n+1)\cdot t/(n+2)$.

Furthermore, by using the equation (II), øDo can be expressed as follows:

$øDo=(n+1)\cdot øDi/n+(n+1)\cdot t/(n+2) \cdots (A)$.

Next, by using the equation (Ib),

$øAo=øDo/(n+1)-øBo$;

therefore, by using the equation (A),

$øAo=øDi/n+t/(n+2)-øBo$,

furthermore, by using the equations (Ia) and (IV),

$øAo=øAi+t/(n+2) \cdots (B)$.

By summarizing the above equations, the outer rotor 120 is formed such that the following equations are satisfied:

$øBo=øbi=øBi \cdots (IV)$;

$øDo=(n+1)\cdot øDi/n+(n+1)\cdot t/(n+2) \cdots (A)$; and

$øAo=øAi+t/(n+2) \cdots (B)$.

FIG. 4 shows the oil pump rotor assembly in which the inner rotor 110 is formed so as to satisfy the above relationship (the diameter øDi of the base circle Di is 52.00 mm, the diameter øAi of the first circumscribed-rolling circle Ai is 2.50 mm, the diameter øBi of the first inscribed-rolling circle Bi is 2.70 mm, and the number of teeth Zi, i.e., "n" is 10), the outer rotor 120 is formed so as to satisfy the above relationship (the outer diameter thereof is 70 mm, the diameter øDo of the base circle Do is 57.31 mm, the diameter øAo of the second circumscribed-rolling circle Ao is 2.51 mm, and the diameter øBo of the second inscribed-rolling circle Bo is 2.70 mm), and the rotors are combined with the clearance "t" of 0.12 mm, and the eccentricity distance "e" of 2.6 mm.

In the casing 150, a suction port having a curved shape (not shown) is formed in a region along which each of the cells C, which are formed between the rotors 110 and 120, moves while gradually increasing the volume thereof, and a discharge port having a curved shape (not shown) is formed in a region along which each of the cells C moves while gradually decreasing the volume thereof.

Each of the cells C draws fluid as the volume thereof increases when the cell C moves over the suction port after the volume of the cell C is minimized in the engagement process between the external teeth 111 and the internal teeth 121, and the cell C discharges fluid as the volume thereof decreases when the cell C moves over the discharge port after the volume of the cell C is maximized.

Note that if the clearance "t" is too small, pressure pulsation is generated in fluid being discharged from the cell C whose volume is decreasing, which leads to generation of cavitation noise, whereby operation noise from the pump is increased. Moreover, the rotors may not smoothly rotate due to the pressure pulsation.

On the other hand, if the clearance "t" is too large, pressure pulsation is not generated, operation noise is decreased, and sliding resistance between the tooth surfaces

is decreases due to a large backlash, whereby mechanical efficiency is improved; however, the fluidtight performance of each of the cells is degraded, and performance of the pump, specifically, the volume efficiency thereof, is degraded. Moreover, because transmission of driving torque in accurately engaged positions is not achieved, and loss in rotation is increased, and finally, mechanical efficiency is degraded.

To prevent the above problems, the clearance "t" is preferably set so as to satisfy the following inequalities:

$0.03 \text{ mm} \leq t \leq 0.30 \text{ mm}$.

In this embodiment, the clearance "t" is set to be 0.12 mm, which is considered to be the most preferable.

In the oil pump rotor assembly formed in a manner such that the above equations (IV), (A), and (B) are satisfied, the profile of the tooth tip of the outer rotor 120 and the profile of the tooth space of the inner rotor 110 have substantially the same shape with respect to each other, as shown in FIG. 5. As a result, as shown in FIG. 5, the circumferential clearances t2 in the engagement phase can be decreased while ensuring the radial clearance t1 such that t/2 is 0.06 mm, which is the same as in conventional rotors; therefore, engagement impacts between the rotors 110 and 120 during rotation are decreased. Furthermore, because the direction along which engagement pressure is transmitted perpendicularly to the tooth surfaces, transmission of torque between the rotors 110 and 120 is performed with high efficiency without slip, and heat generation and noise due to sliding resistance can be reduced.

In this embodiment, as in the first embodiment, when a clearance, which is defined between the teeth of the inner and outer rotors 110 and 120 that together form one of the cells which has the minimum volume among the cells, is designated as "a", a clearance, which is defined between the teeth of the inner and outer rotors 110 and 120

that together form one of the cells whose volume is increasing during rotation of the inner and outer rotors 110 and 120, is designated as "b", and a clearance, which is defined between the teeth of the inner and outer rotors 110 and 120 that together form one of the cells which has the maximum volume among the cells, is designated as "c" (clearances "a", "b", and "c" are not shown), the following inequalities are satisfied:

$a \leq b \leq c$, and $a < c$.

Moreover, when the clearance "b" of the cell positioned forward as viewed in the direction of rotation is further designated as "b1", and the clearance "b" in the cell positioned backward as viewed in the direction of rotation is further designated as "b2", the following inequality is satisfied:

$b1 \leq b2$.

Furthermore, when a clearance, which is defined between the teeth of the inner and outer rotors 110 and 120 that together form one of the cells whose volume is decreasing during rotation of the inner and outer rotors 110 and 120, is designated as "d", the following inequalities are satisfied:

$a \leq b \leq c$, $a < c$, and $a \leq d \leq c$.

Moreover, when the clearance "d" in the cell positioned backward as viewed in the direction of rotation is further designated as "d1", and the clearance "d" in the cell positioned forward as viewed in the direction of rotation is further designated as "d2", the following inequality is satisfied:

$d1 \geq d2$.

FIG. 6 is a graph showing comparison between noise from a pump incorporating a conventional oil pump rotor assembly and noise from another pump incorporating the oil pump rotor assembly according to the present embodiment. According to the graph, noise from the oil pump incorporating the oil pump rotor assembly according to the

present embodiment is less than that of the conventional oil pump rotor assembly, i.e., the oil pump rotor assembly of the present embodiment is quieter.

Next, a third embodiment of the present invention will be explained below with reference to FIGS. 7 to 10.

The oil pump rotor assembly shown in FIG. 7 includes an inner rotor 210 provided with "n" external teeth ("n" indicates a natural number, and n=10 in this embodiment), and an outer rotor 220 provided with "n+1" internal teeth (n+1=11 in this embodiment) which are engageable with the external teeth. The inner rotor 210 and the outer rotor 220 are accommodated in a casing 250.

Between the tooth surfaces of the inner rotor 210 and outer rotor 220, there are formed a plurality of cells C in the direction of rotation of the inner rotor 210 and outer rotor 220. Each of the cells C is delimited at a front portion and at a rear portion as viewed in the direction of rotation of the inner rotor 210 and outer rotor 220 by contact regions between the external teeth 211 of the inner rotor 210 and the internal teeth 221 of the outer rotor 220, and is also delimited at either side portions by the casing 250, so that an independent fluid conveying chamber is formed. Each of the cells C moves while the inner rotor 210 and outer rotor 220 rotate, and the volume of each of the cells C cyclically increases and decreases so as to complete one cycle in a rotation.

The inner rotor 210 is mounted on a rotational axis so as to be rotatable about an axis Oi. Each of the tooth profiles of the inner rotor 210 is formed such that the tooth tip profile thereof is formed using an epicycloid curve which is formed by rolling a first circumscribed-rolling circle Di along a base circle "bi" of the inner rotor 210 without slip, and the tooth space profile thereof is formed using a hypocycloid curve which is formed by rolling a first inscribed-rolling circle "di" along the base circle "bi" without slip.

The outer rotor 220 is mounted so as to be rotatable, in the casing 250, about an axis Oo which is disposed so as to have an offset (the eccentricity distance is "e") from the axis Oi. Each of the tooth profiles of the outer rotor 220 is formed such that the tooth space profile thereof is formed using an epicycloid curve which is formed by rolling a second circumscribed-rolling circle Do along a base circle "bo" of the outer rotor 220 without slip, and the tooth tip profile thereof is formed using a hypocycloid curve which is formed by rolling a second inscribed-rolling circle "do" along the base circle "bo" without slip.

When the diameter of the base circle "bi" of the inner rotor 210, the diameter of the first circumscribed-rolling circle Di, the diameter of the first inscribed-rolling circle "di", the diameter of the base circle "bo" of the outer rotor 220, the diameter of the second circumscribed-rolling circle Do, and the diameter of the second inscribed-rolling circle "do" are assumed to be øbi, øDi, ødi, øbo, øDo, and ødo, respectively, the equations which will be discussed below must be satisfied between the inner rotor 210 and the outer rotor 220. Note that dimensions will be expressed in millimeters.

First, with regard to the inner rotor 210, because the total of a multiple of the rolling distance of the first circumscribed-rolling circle Di and a multiple of the rolling distance of the first inscribed-rolling circle "di" must agree with the length of circumference of a base circle, i.e., the length of circumference of the base circle "bi" of the inner rotor 210 must be equal to the length obtained by multiplying the sum of the rolling distance per revolution of the first circumscribed-rolling circle Di and the rolling distance of the first inscribed-rolling circle "di" by an integer (i.e., by the number of teeth of the inner rotor 210),

$\pi \cdot øbi = n \cdot \pi \cdot (øDi + ødi)$, i.e.,

$øbi = n \cdot (øDi + ødi) \cdots$ (Ia).

Similarly, with regard to outer rotor 220, the length of circumference of the base circle "bo" of the outer rotor 220 must be equal to the length obtained by multiplying the sum of the rolling distance per revolution of the second circumscribed-rolling circle Do and the rolling distance of the second inscribed-rolling circle "do" by an integer (i.e., by the number of teeth of the outer rotor 220),

$\pi \cdot øbo=(n+1) \cdot \pi \cdot (øDo+ødo)$, i.e.,

$øbo=(n+1) \cdot (øDo+ødo)$ ··· (Ib).

The tooth tip profile of the inner rotor which is formed by the first circumscribed-rolling circle Di with respect the tooth space profile of the outer rotor which is formed by the second circumscribed-rolling circle Do, and the tooth tip profile of the outer rotor which is formed by the second inscribed-rolling circle "do" with respect the tooth space profile of the inner rotor which is formed by the first inscribed-rolling circle "di" are determined such that the following inequalities are satisfied:

$øDo > øDi$; and

$ødi > ødo$,

so that a large backlash which is defined between the tooth surfaces of the rotors during engagement is ensured. Here, the backlash is a gap formed between the tooth surface of the inner rotor, which is opposite to the tooth surface to which force is applied during engagement, and the tooth surface of the outer rotor.

Moreover, because the inner rotor and the outer rotor engage each other, one of the following equations must be satisfied:

$øDi+ødi=2e$; and

$øDo+ødo=2e$.

Furthermore, in this invention, in order to make the inner rotor 210 smoothly

rotate in the outer rotor 220 while ensuring a tip clearance and an appropriate size of backlash, and reducing an engagement resistance, the diameter of the base circle "bo" of the outer rotor 220 is made greater than that in a conventional case so that the base circle "bi" of the inner rotor 210 does not contact the base circle "bo" of the outer rotor 220 at the engagement region at which the inner rotor 210 engages the outer rotor 220, i.e., the following inequality is satisfied:

$(n+1) \cdot øbi < n \cdot øbo$.

Using this inequality, and the equations (Ia) and (Ib), the following inequality is derived:

$(øDi+ødi) < (øDo+ødo)$.

The above-mentioned engagement region is a region at which the tooth tip of one of the internal teeth 221 of the outer rotor 220 directly faces one of the tooth spaces between the external teeth 211 of the inner rotor 210.

The inner rotor 210 and the outer rotor 220 are formed such that the following inequalities are satisfied:

$0.005\ mm \leq (øDo+ødo)-(øDi+ødi) \leq 0.070\ mm$ (hereinafter, $(øDo+ødo)-(øDi+ødi)$ is simply designated as "A").

In the embodiment, the inner rotor 210 (the diameter øbi of the base circle is 65.00 mm, the diameter øDi of the first circumscribed-rolling circle Di is 3.90 mm, the diameter ødi of the first inscribed-rolling circle "di" is 2.60 mm, and the number of teeth "n" is 10), and the outer rotor 220 (the outer diameter thereof is 87.0 mm, the diameter øbo of the base circle "bo" is 71.599 mm, the diameter øDo of the second circumscribed-rolling circle Do is 3.9135 mm, and the diameter ødo of the second inscribed-rolling circle "do" is 2.5955 mm), each of which is formed so as to satisfy the above-mentioned conditions, are combined with an eccentricity distance "e" of 3.25 mm

to form the oil pump rotor assembly. In this embodiment, the width of the teeth of the rotors (the size in the direction of the rotational axis) is set to be 10 mm. Because the diameter ødi of the first inscribed-rolling circle "di" is set to be 2.60 mm, the diameter øDo of the second circumscribed-rolling circle Do is set to be 3.9135 mm, and the diameter ødo of the second inscribed-rolling circle "do" is set to be 2.5955 mm, "A" is 0.009 mm (refer to FIG. 8).

In the casing 250, a suction port having a curved shape (not shown) is formed in a region along which each of the cells C, which are formed between the rotors 210 and 220, moves while gradually increasing the volume thereof, and a discharge port having a curved shape (not shown) is formed in a region along which each of the cells C moves while gradually decreasing the volume thereof.

Each of the cells C draws fluid as the volume thereof increases when the cell C moves over the suction port after the volume of the cell C is minimized in the engagement process between the external teeth 211 and the internal teeth 221, and the cell C discharges fluid as the volume thereof decreases when the cell C moves over the discharge port after the volume of the cell C is maximized.

If "A" is too small, the tip clearance and the backlash cannot be appropriately set, and engagement noise between the external teeth 211 of the inner rotor and the internal teeth 221 of the outer rotor cannot be reduced.

If "A" is too great, the difference between the height (the size of a tooth along the normal of the base circle) of the external teeth 211 of the inner rotor and the height of the internal teeth 221 of the outer rotor, and the difference between the width (the size of a tooth along the circumference of the base circle) of the external teeth 211 of the inner rotor and the width of the internal teeth 221 of the outer rotor cannot be appropriately set; therefore, the backlash may become zero at some regions during the rotation of the inner

and outer rotors 210 and 220. In this case, the rotors cannot smoothly rotate; therefore, mechanical efficiency may be degraded, and excessive noise may be emitted due to impacts between the external teeth 211 and the internal teeth 221.

Accordingly, it is preferable that "A" be set in the range from 0.005 mm to 0.070 mm, and in this embodiment, "A" is set to be 0.009 mm.

In the oil pump rotor assembly configured as explained above, the tooth tip profile of the outer rotor 220 substantially coincides with the tooth space profile of the inner rotor 210. As a result, as shown in FIG. 8, the circumferential clearances "ts" along the base circle are made small while the tip clearance "tt" is maintained as in a conventional case; therefore, the impacts applied to the rotors 210 and 220 during rotation become small. Accordingly, impacts between the internal teeth 221 of the outer rotor 220 and external teeth 211 of the inner rotor 210 can be prevented even when driving torque for the oil pump rotor assembly changes while oil pressure in the oil pump rotor assembly is low; therefore, quietness of the oil pump rotor assembly can be ensured. Moreover, because the rotational force is transmitted in the direction substantially perpendicular to the tooth surfaces, torque is transmitted between the rotors 210 and 220 without slip and with high efficiency, heat generation and noise due to sliding friction can be reduced.

FIG. 9 is a graph showing comparison between a backlash (shown by a broken line in FIG. 9) of a conventional oil pump rotor assembly with respect to the rotational position of the inner rotor and a backlash (shown by a solid line in FIG. 9) of the oil pump rotor assembly of the present embodiment with respect to the rotational position of the inner rotor. According to this graph, in the oil pump rotor assembly of the present embodiment, the backlash in the engagement region, the backlash in the process in which the volume of the cell C increases, and the backlash in the process in which the volume

of the cell C decreases, are smaller than those in the conventional oil pump rotor assembly, and the backlash at a position at which the volume of the cell C is maximized is substantially equal to that in the conventional oil pump rotor assembly. Accordingly, in the oil pump rotor assembly of the present embodiment, because the fluidtight performance of the cell C having the maximum volume can be ensured, and fluid conveying efficiency can be maintained substantially the same as in a conventional pump. In FIG. 9, only the backlash at a rotational position of the inner rotor from 0° to 198° is shown because the backlash at a rotational position of the inner rotor from 198° to 396° is similar (symmetrical) to that from 198° to 0° shown in FIG. 9.

FIG. 10 is a graph showing comparison between noise from an oil pump incorporating a conventional oil pump rotor assembly and noise from the oil pump incorporating the oil pump rotor assembly of the present embodiment. According to this graph, the oil pump rotor assembly of the present embodiment makes it possible to reduce noise when compared with the conventional oil pump rotor assembly, i.e., a quiet oil pump can be made, because the backlash in the engagement region, the backlash in the process in which the volume of the cell C increases, and the backlash in the process in which the volume of the cell C decreases, are smaller than those in the conventional oil pump rotor assembly as shown in FIG. 9.

The various elements, dimensions thereof, and combinations thereof explained in the above embodiments are merely examples, and various modifications may be made in accordance with design requirements without departing from the scope of the present invention.

For example, in the above embodiments, the rotors that form the internal gear type oil pump rotor assembly are so-called cycloid rotors having teeth which are formed

using cycloid curves; however, any rotors may be used which satisfy the above-mentioned clearance conditions, such as so-called trochoid rotors which includes an inner rotor having teeth which are formed using a trochoid envelope curve which is formed by moving a trajectory circle, whose center is positioned on a trochoid curve, along the trochoid curve, and an outer rotor that is engageable with the inner rotor.

INDUSTRIAL APPLICABILITY

As explained above, according to the internal gear type oil pump rotor assembly of the present invention, because the clearance between the rotors that together form the cell is minimized at an engagement region, and then the clearance is continuously increased, without decreasing, to a maximum size, backlash at a position at which the teeth engage each other is minimized, and a sufficient clearance is ensured at a rotational position at which the teeth do not contribute to engagement.

According to another internal gear type oil pump rotor assembly of the present invention, because the clearance between the rotors that together form the cell is maximized, and then the clearance is continuously decreased, without increasing, to a minimum size at the engagement region, backlash at a position at which the teeth engage each other is minimized, and a sufficient clearance is ensured at a rotational position at which the teeth do not contribute to engagement.

Accordingly, the external teeth engage the internal teeth at a position at which a slip component is minimized so as to transmit rotational force, and the external teeth and the internal teeth do not contribute to transmitting rotational force at a position at which a slip component is increased. Therefore, an internal gear type oil pump rotor assembly can be obtained which does not emit excessive noise while having low levels of friction and high mechanical efficiency.

According to another internal gear type oil pump rotor assembly of the present invention, a cycloid type rotor assembly which is formed using cycloid curves and a trochoid type rotor assembly which is formed using trochoid curves, both of which have been conventionally used, can be made so as to emit less noise and to have lower level of friction; therefore, an internal gear type oil pump having high performance can be obtained.